Volume 8 March, 1922 *Number 3*

Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

A 1,000 Horse Power Submarine Diesel Engine

PUBLISHED MONTHLY BY
THE TEXAS COMPANY, U.S.A.
TEXACO PETROLEUM PRODUCTS

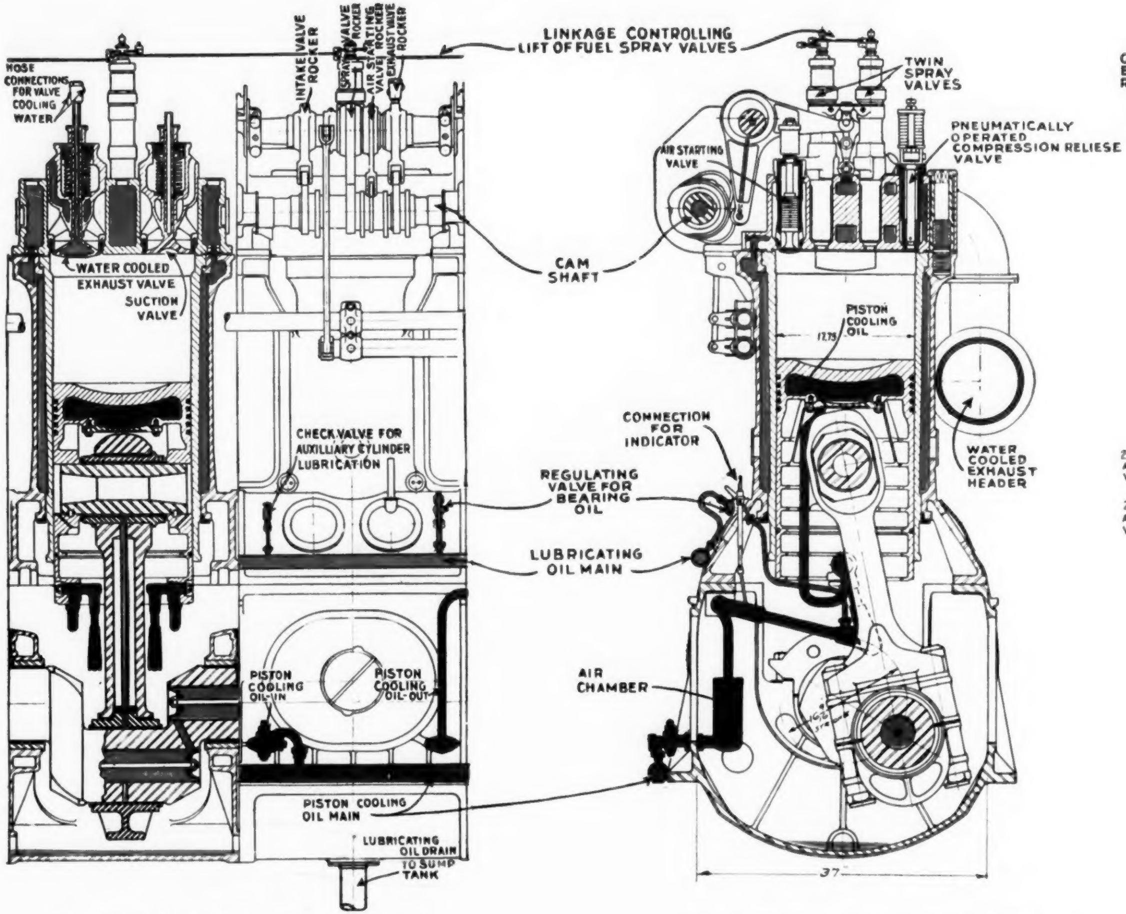

Fig. 1—Longitudinal Section and Elevation of 2 Main Cylinders.

Fig. 2—Cross Section thru a Main Cylinder.

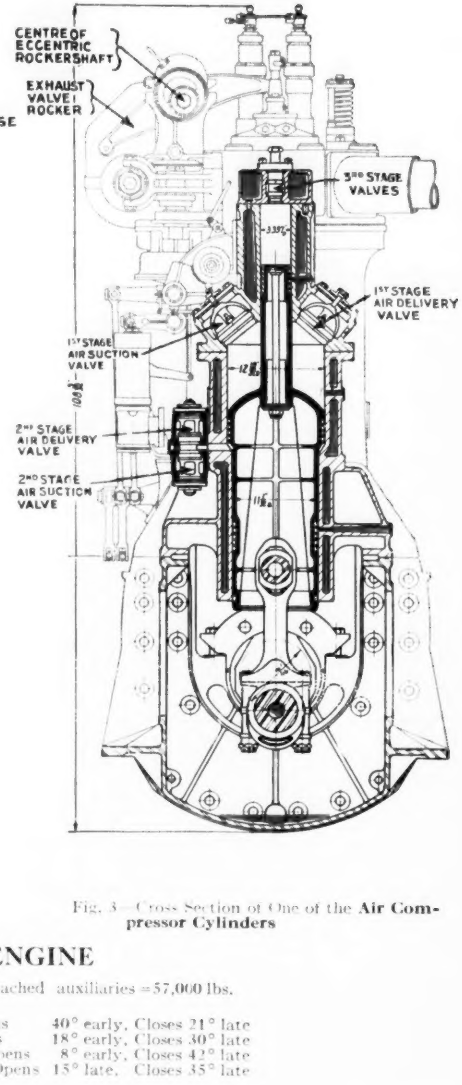

Fig. 3—Cross Section of One of the **Air Compressor Cylinders**

Sections of 1000 B.H.P. 6 Cylinder SUBMARINE DIESEL ENGINE

Rated at 420 r.p.m.—Overload = 1,200 B.H.P. at 450 r.p.m. Bore = $17\frac{3}{4}$" Stroke = $16\frac{9}{16}$". Weight with attached auxiliaries = 57,000 lbs.

Colors:	Green = Water
	Red = Lubricating Oil
	Brown = Piston Cooling Oil

Valve Timing:		
	Exhaust Opens	40° early, Closes 21° late
	Suction Opens	18° early, Closes 30° late
	Fuel Spray Opens	8° early, Closes 42° late
	Starting Air Opens	15° late, Closes 35° late

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Published Monthly by

The Texas Company, 17 Battery Place, New York City

Vol. VIII — March, 1922 — No. 3

Change of Address: In reporting change of address kindly give both old and new addresses.

A 1000 Horse Power Submarine Diesel Engine

THE engines used by the Germans to propel their submarines are probably the most highly developed Diesel Engines ever manufactured in quantity. About 500 of the 1,200 H.P. type were used in their 800-ton submarines, besides many of greater power for larger boats, including the 3,000 H.P. type used on the 2,700 ton submarines. These engines, as well as smaller types, are all somewhat similar as to general and detailed design; they are all 4 cycle single-acting air-injection engines. The extraordinary reliability and flexibility of these engines were a source of considerable surprise to those present at the time these submarines were surrendered. It was noted that the engines ran smoothly at but 1-10 of their maximum speed, and even while idling with no load at this low speed they had a clear smokeless exhaust. To still further emphasize the performance of these engines, a comparison with ordinary Diesel engines shows that the power developed for a given size cylinder is over twice as great for the former. This means that twice as much heat is produced in a given space with the result that the exhaust temperature is about 1,000° F. while ordinary Diesel engines have but 600° to 700°. This difference in temperature greatly increases the mechanical difficulties resulting from heat which has necessitated the use of oil cooled pistons as well as water cooled exhaust valves and exhaust valve cages.

Engines of similar design are being manufactured at the Navy Yard, New York, the first having recently completed successful shop trials. The Navy Department has been kind enough to give us access to drawings, photographs and test data of these engines, which have been made use of in preparing this article. It is so seldom possible to publish much of the minute technical details responsible for the success of such a perfected mechanism, that this data has been given for those who can make use of it. Much of this information is being published here for the first time.

It must be borne in mind, however, that submarine engines are so highly specialized for their particular service that they can hardly be considered as commercial types particularly on account of their higher cost per horse power. Several of these engines have nevertheless been installed in the hulls of merchant ships, and even though their r.p.m. has been reduced somewhat it has been too high for the propellers of these ships and reduction gearing has been used. The use of engines with such high power development for their size will be followed with intense interest by those who believe that

reliability is not necessarily sacrificed by the high power development and speed, as well as by the comparatively light weight of these engines.

As perfection of the details of the lubricating system has had much to do with the extraordinary success of these engines, particular study is given in this article, to these details as well as to the handling and control of the fuel. It is not possible here to go into the interesting and ingenious mechanical details concerned with their construction, though much can be learned from the sectional drawings shown in Figs. 1, 2 and 3.

Main Engine Lubricating System

Every moving part of the engine except the valve stems and part of the maneuvering gear. is lubricated from a full pressure circulating system which reaches even the rollers on the valve rockers. The completeness of this circulating system reminds one of the lubricating systems developed especially for airplane engines, and as will be seen later, the bearing clearances and grooving are also similar to what is considered as best airplane and racing engine practice. The few parts not lubricated by the circulating system are supplied with oil

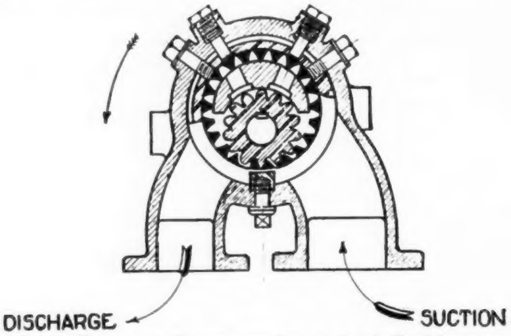

Fig. 4.—The rotary **oil pump** used as an electrically driven **auxiliary** to provide cooling and lubricating oil when the engine is not running. At 1,800 r.p.m. it delivers 156 gallons a minute. The ring gear is only $4\frac{1}{4}''$ inside diameter, the pinion about $3\frac{9}{16}''$ outside diameter, and the $5\frac{5}{16}''$ length of the teeth is $3\frac{1}{2}''$.

and grease cups, which are sufficient since the maneuvering gear does not require much oil. Even if the lubrication of these parts is neglected there will be no other ill effect than making the work of the operator greater while maneuvering, which is assurance that he will not neglect it. Probably the desire to avoid further complication from additional oil piping is the reason why these parts are not included in the circulating system. As it is, over 200 feet of oil piping are used, varying in size from $\frac{1}{4}''$ O. D. to $4\frac{1}{4}''$ O. D. Operators are instructed to oil by hand every hour the valve stems and all parts not included in the circulating system. A point of interest in regard to the valve lubrication is that the stems are intended to have sufficient clearance in their guides so that there is no contact at this point and therefore no oil is necessary. The only part lubricated is the piston-like guide above the spring, which is not exposed to heat.

A main oil sump tank of about 500 gallons is used for storage, and as it is located below

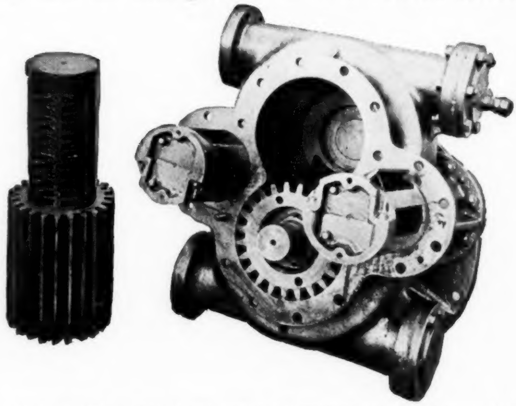

Fig. 5.—The direct driven **main oil pump** partly dis-assembled, showing the cages carrying the double check valves by which oil is pumped in one direction only, irrespective of the direction of engine rotation. The idler gear is shown with its spindle partly withdrawn to show the ports by which oil trapped between engaging teeth, is by-passed through holes drilled between the gear teeth. The gears are $5\frac{3}{4}''$ outside diameter, 7″ long and have 21 teeth. At 400 r.p.m. it delivers 55 gallons a minute.

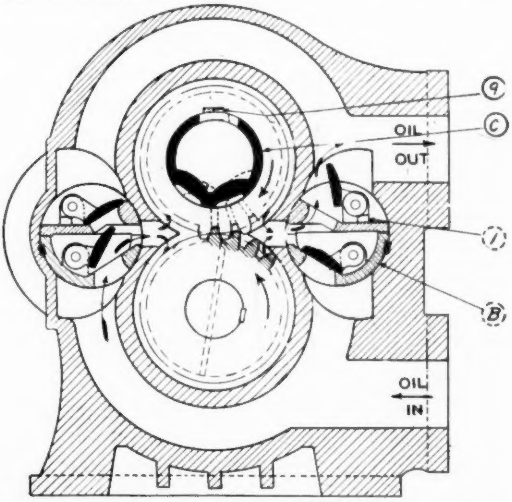

Fig. 6.—Sectional drawing of the **main oil pump** showing the constructional details of Fig. 5

(c) is the idler gear spindle in which the by-passing ports are cut.

(9) is the spindle key which is narrower than its key-way and allows the spindle to change the port position when the pump rotation is reversed.

the engine all the drainage of oil flows back to it by gravity, through outlets at both ends of the crankcase. Operators are instructed not only to clean the tank every 8 or 10 weeks and the strainer contained in it every 50 hours of service, but to blow down the whole oiling system with steam once a year. From this tank the oil is drawn by either an auxiliary electrically-driven oil pump or the main engine driven oil pump. A "stop-lift" check valve on the suction line near the tank makes it unnecessary to prime the oil pumps when starting. Before starting the engine, complete lubrication is assured by the auxiliary pump which runs in parallel with the main oil pump. This auxiliary pump, shown in Fig. 4, is an interesting rotary type. It consists of a single spur gear mounted on ball bearings, and running in mesh with an internal gear. The outer diameter of the internal gear is turned down so as to break through to the space between the teeth, like a squirrel cage, giving passage for oil except for flanges at each end which hold the teeth together.

The main pump shown in Figs. 5 and 6, is a gear pump driven at crank-shaft speed by spur gearing from the vertical shaft which drives the camshaft, and operates with the gears in a horizontal plane. The two principal features of this pump are: (A) The means taken to release oil trapped between the teeth as they mesh, thus making it operate quietly, and (B) The check valves which allow oil to flow in one direction irrespective of the direction of pump rotation. The first feature is accomplished by means of nine 9-32″ holes drilled through the idler gear radially between the teeth. The spindle upon which this gear rotates is of comparatively large diameter, and has a series of transverse ports milled tangentially as shown in Fig. 5. Oil in the space between two teeth running into mesh can pass into these ports and back to the discharge side of the pump through similar holes between teeth which are still free. In a similar way the teeth going out of mesh are supplied with oil from the suction side. This results in eliminating the shocks and side pressure caused by teeth closing in on a trapped slug of oil—as well as the partial vacuum on the suction side. Thus the efficiency of the pump is raised, its life is prolonged, and the operation is made unusually quiet.

When the pump rotation is reversed by a reversal of the engine, a slight change is necessary in the location of the ports. This is automatically accomplished by the friction between the idler and its spindle, which tends to drag the spindle in the same direction that the gear rotates. A keyway in the pump housing, which is wider than the key in the spindle, allows a few degrees of rotation of the spindle when the gear motion is reversed, and thus secures the proper setting of the spindle ports. The Naval Officers who have had experience with the operation of this pump are enthusiastic in their praise of it.

From the pump, oil flows through a $2\frac{3}{8}''$ I. D. pipe to the filter shown in Fig. 7. This is of the duplex type, and of such capacity that, by the three-way valve, either half may be by-passed for cleaning while the other is in use. In normal operation both sides are used in parallel. Instructions recommend that these filters be cleaned every 12 to 24 hours of service. Attention is called to the arrangement of the six discs with gauze on each side, by which a large area is secured in a small space.

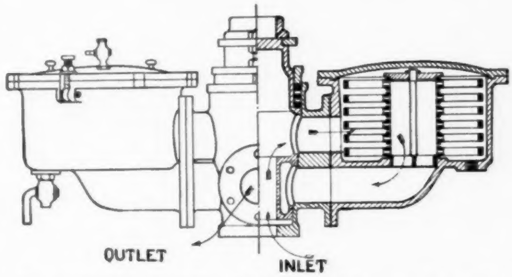

Fig. 7.—The duplex **oil filter** with its special three-way valve for by-passing one half while the other is being cleaned. The arrangement of the wire gauze disks makes it very compact. Each half has a total of 576 square inches of 36 mesh No. 32 wire screen. The maximum pressure drop allowed through it is $4\frac{1}{2}$ lbs. per square inch.

The filtered oil passes next to an oil cooler shown in Fig. 8. It will be noticed that the water tubes and the baffles for the oil are suspended as a unit from the upper header, while the lower header is free, its vertical position being located by the tubes only. This results in two distinct advantages: (1) The tubes are free to expand and contract without inducing stresses between themselves and the casing; (2) The tube unit can be removed

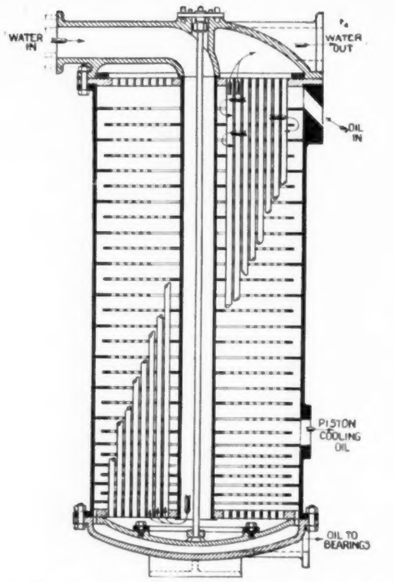

Fig. 8.—Oil passes through the **oil cooler** from top to bottom in an alternately contracting and expanding stream by means of baffles, while the water enters at the top, passes down a central 3″ tube to the lower header from which it returns to the top through 462 copper tubes $\frac{5}{8}$″ O.D. and $42\frac{1}{2}$″ long. There is a cooling surface of 280 square feet, and a circulation of water at 76° F. cools the 45 gallons per minute of piston cooling oil from 126° F. to 104°, and the remaining 16 gallons per minute of bearing oil, to 90° F. This is equivalent to a heat transfer rate of about 38 B.T.U.'s per square foot per degree per hour. The cooler holds 32 gallons of oil. The 218 gallons per minute of cooling water for the engine passes through this cooler, after which it is divided into three streams (1) about 22 gallons per minute to the air compressor cylinders; (2) 62 gallons per minute to the intercoolers for the air compressor; and (3) about 134 gallons per minute to the main cylinder jackets.

readily from the casing, making cleaning and inspection a simple matter. Zinc rings are bolted to the top bronze header and the bottom iron cover to reduce corrosion from the salt water.

At the cooler, oil is divided into two separate systems, one for piston cooling and the other for bearing lubrication. The oil for the piston cooling system is drawn from a point $8\frac{3}{8}$″ from the bottom, but the oil for the bearings is cooled still further by passing to the bottom between baffles which, on account of the reduced volume of oil, are spaced about $\frac{7}{8}$″ apart instead of $1\frac{1}{2}$″ as in the upper section. These two oiling systems will now be traced in detail.

Piston Cooling System

Oil for piston cooling passes from the cooler to a header along the base of the crankcase, through air chambers and then to the pistons by pairs of hinged pipes, as can best be seen from Figs. 1 and 2. There are, of course, two of these oil connections to each piston, one leading the oil in and the other out. The purpose of air chambers is to cushion the "water-hammer" effect of the reciprocating mass of oil in the piston, which is severe enough to endanger tight oil joints.

The hinged oil pipes fulcrum on two pins at the lower end of each piston, one on each side. From these pins, which are hollow, oil is led by other piping to and from the closed-in pocket under the piston head as shown in Fig. 9. No precautions are taken to remove air from this pocket when starting, for in operation if any air is present the oil is splashed so violently that it is sure to be removed with the oil in a few minutes as a sort of froth. Oil from the pistons is led through $\frac{13}{16}$″ I. D. tubes to a point near the operator's station, where the flow from each piston can be observed as it is collected in a funnel and returned to the sump tank. Thermometers are placed near each outlet, where the uniformity of the piston

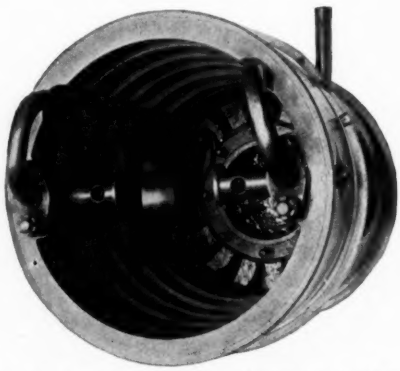

Fig. 9.—Interior of a **main piston** showing the oil piping to the oil pocket in the piston head, and the pin joints through which oil connections are made to stationary parts of the engine. One of the taper pins which locks the wrist pin home, is shown partly withdrawn. The piston clearances, cold, are: above the top ring, from $\frac{1}{64}$″ over to $\frac{1}{64}$″ under $\frac{1}{8}$″; decreasing to 0.021″ just below the upper wiper ring, and to 0.017″ at the bottom of the piston skirt.

cooling can be observed. This oil cooling system is very satisfactory, and has a double advantage over use of water for the same purpose; there is no danger of incrustions of salt in the cooling system, and any leakage into the crankcase will not cause trouble with emulsions. It is true that a given rate of oil flow will carry off less than one-half as much heat as water, but even in this engine all the heat is carried off that is necessary, with an oil temperature no higher than about 130° F. Approximately 280 B.T.U.'s per hour are

Fig. 10.—**Hinged piping connections** between the crankcase and piston for the piston cooling oil. Their location in the engine is shown in Fig. 2. The two vertical pipes at the right are shown with the pins in place, by which connections are made to a piston. These pins are the same as those shown in Fig. 9. Below are shown the air chambers bolted to the crankcase, the one at the left being for the incoming oil, and the other for the outgoing oil.

carried off per I. H. P. of the engine. It is interesting to compare this method of conducting oil to and from the pistons, with that of Fig. 11 which is used by Sulzer and many others.

Bearing Lubrication System

The second branch of the oil system from the cooler, as has been mentioned, is for lubrication. Oil is led through a 1½" I. D. pipe to a pressure reducing valve shown in Fig. 12. Mounted with this reducing valve is a pressure relief valve which limits the pressure in the whole system back to the pump, and hence the piston cooling system also. From this reducing valve, oil is supplied to the lubricating oil header which runs along the crankcase, from which ½" I. D. leads pass to the center of each of the main crankshaft bearings at the top. Control over the flow to each bearing is secured by special stop cocks which can be locked in any one of several positions intermediate between full open and shut, by means of a pin in the handle which may be dropped into various holes in a segment fixed to the valve body. Occasional touching of the bearings with the hand will indicate by their temperature if they are running with too much or too little oil. A noteworthy feature of the crankshaft bearings is the absence of criss-cross oil grooves in the pressure surfaces. There is only one circumferential groove by means of which oil is led to drilled passages in the crank-shaft and thus to the crankpins. All holes in the crankpins and bearings are threaded to take plugs when a shaft is disassembled for shipping, so that dirt cannot get inside.

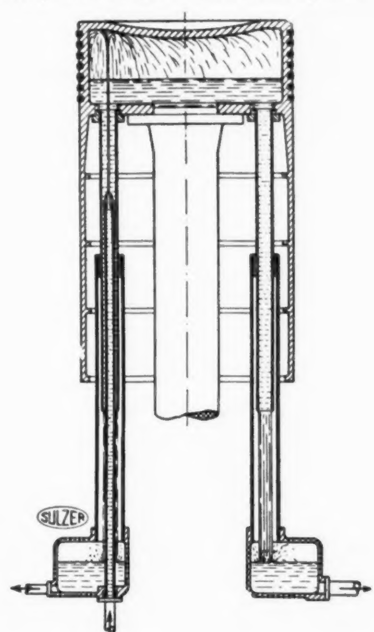

Fig. 11.—The piston cooling system used by many Diesel engine manufacturers, in which telescopic tubes are used instead of hinged joints.

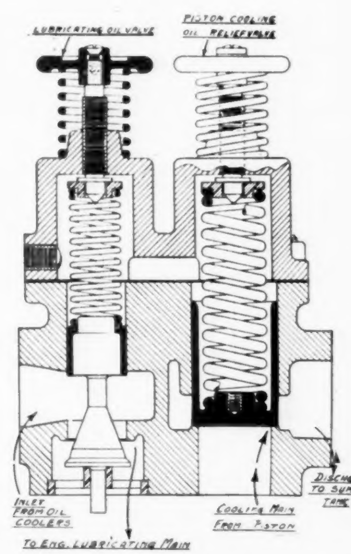

Fig. 12.—**Oil pressure regulating** and **relief valve.** By means of the pressure reducing valve at the left, the oil pressure on the bearing is reduced to about 20 lbs. per square inch at 200 r.p.m., and increases to a maximum of 30 lbs. at 450 r.p.m. of the engine. The right hand member is set to protect the oil system from any pressure above 50 lbs.

From the crankpin bearing, oil is led up to the wrist pin bearing by a ½" I. D. tube through the hollow connecting rod. The wrist pin bearing is a two part steel forging which, contrary to usual practice, is lined with babbitt and grooved as shown in Fig. 14. The wrist-pin, where it bears on the bearing, is of case-hardened steel, and each end is slightly tapered so it can be fitted solidly in the piston. It is held in place by taper pins which are driven home, these being further locked by cotters,

Fig. 13.—A disassembled **connecting rod bearing** showing in particular the type of grooving, which is $\frac{3}{32}$"x⅜" wide. The relieved areas at the side are 3"x6½" long, and 0.006" deep. Laminated shims by which the diametrical clearance is adjusted to 0.006" are shown held to the lower half (at the left) by small screws. By means of the shim at the right, the effective length of the connecting rod is adjusted for the compression pressure desired (450 lbs. per square inch). This bearing is just over 9$\frac{7}{16}$" diameter, and is but 9½" long, though the crank-pin length is 11$\frac{1}{32}$". The resulting large side clearance gives abundant room for a large radius fillet. The crankshaft bearings are almost identical, except 1⅜" shorter, and have $\frac{3}{32}$" greater side clearance. The longitudinal position of the shaft is fixed by thrust collars on the rear end bearing.

making a very simple and secure fastening. The assembly of this bearing in the connecting rod is unusual. The lower half is first put in place, where it is secured by the flanges on each end when the upper half is pressed into place. To make this possible one of the flanges of the upper half is cut down lower than the thickness of the shim which goes between the rod and the top "brass," the exact thickness of which is selected to require a force of one ton to press this part into place.

Cylinder Lubrication

Oil thrown from the crankpin bearings is depended upon to supply the principal cylinder lubrication. An additional means is provided by leading oil directly from the lubricating oil header to points on opposite sides of each cylinder. A check valve on the line to each cylinder normally prevents any flow of oil, and is so located as to be convenient to the operator who presses each one open for a moment when the engine is started, or at any time when he may think one of the cylinders may need additional oil. In order to prevent too much oil from passing up into the combustion space, no other precaution is taken than the use of two special oil wiper rings, one of them being the lowest of the five rings at the upper end of the piston. These two rings which, like the others, are ½" square in section, are beveled on the outside from the top to within ⅛" of the bottom, producing a clearance of $\frac{3}{64}$" at the top. This concentrates the ring pressure on a width of ⅛" when the ring is new. As the ring wears, the width of contact increases until it is ¼", at which time the ring should be replaced by a new one. Below the grooves for these rings, the piston is chamferred for ½" to a depth of $\frac{1}{16}$". Six ⅛" holes are drilled through the piston walls at these chamferred spaces, to return the oil wiped from the cylinder wall. Drain holes are drilled also to ends of the wrist pin spaces.

The cam-shaft and valve rocker shaft have internal oil passages for their entire length, and are supplied by oil from the same header that feeds the main bearings and auxiliary cylinder lubricators. Holes through the rocker shaft where it passes through the rockers, supply oil to the latter and to the cam rollers by tubes which lead oil from the rockers to the hollow studs on which the rollers turn. The cam-shaft bearings get oil in a similar way from holes in the cam-shaft. The two sets of spiral gears through which the cam-shaft is driven by a vertical shaft at the rear of the engine, are lubricated by streams of oil which strike the gears where they run together. The gears do not run submerged in oil. These oil streams are fed by the same header which supplies the main bearings, the camshaft bearings and the valve rockers and rollers.

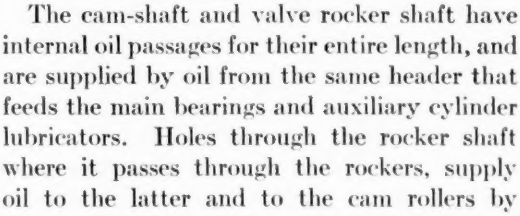

Fig. 14.—The **connecting rod** and babbitt lined **wrist-pin bearing** showing the very small amount of grooving used. There are four crosses like the one shown, at 90° from each other and connected together by a single groove. The bearing is 11" long and 6½" diameter, and has a radial clearance between 0.004" and 0.005". As the side clearance is only $\frac{1}{32}$" the connecting rod is thus located laterally by this bearing instead of by the crank-pin bearing.

Thus all journals of the engine proper, which rotate while the engine is running, are supplied by a circulation of cooled oil under pressure, the pressure increasing with the engine speed.

Air Compressor

The air compressor has four stages arranged in a unique way, there being two "differential" pistons driven at 180° from each other by cranks on an extension of the engine crankshaft. Each of these pistons has a 1st and 2nd stage—operating in parallel. The aft piston also handles the 3rd stage, while the forward piston handles the 4th stage. Fig. 3 shows a section through the forward piston and its water jacketed cylinders, and Fig. 15 shows the piston itself. Notice that the bottom of the piston has to seal the 2nd stage from leakage of air into the crankcase. This minimizes the tendency for too much oil to work up into the cylinder from the crankpin, for there is not only no suction to draw it up but if there is any leakage of air it will tend to force the oil down from the cylinder walls.

This particular arrangement of the different stages on one piston, in which the 2nd stage acts on the down stroke while the 1st and 3rd act on the up stroke, instead of all three acting in one direction, not only reduces the bearing pressures but actually reverses them. It makes their lubrication much simpler than if the pressure were always in the same direction. Dividing the work of the first two stages between two cylinders in parallel not only insures a fairly steady stream of air, but still further reduces the total bearing pressure. The crankpin and wrist-pin bearings are lubricated from the hollow engine shaft by the same circulating system used for the other crankshaft bearings.

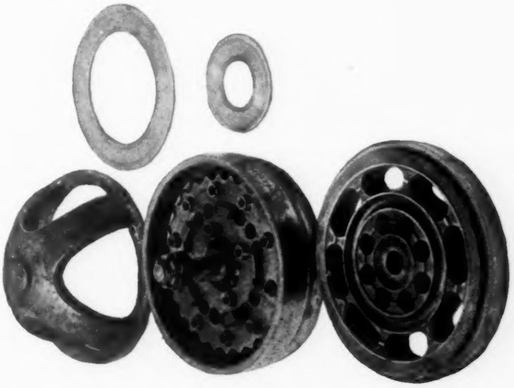

Fig. 16.—**Automatic suction valve** for the first stage. The valve itself consists of the two thin ground spring steel rings shown below. The seat is at the right, while at the center is shown the 14 small springs which normally hold the valves on their seats.

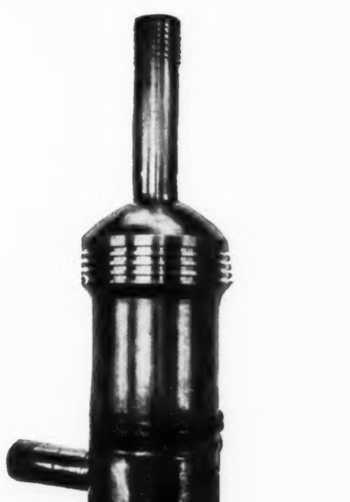

Fig. 15.—**Air compressor piston**, with the wrist-pin partly withdrawn. The area above the largest diameter is for the 1st stage, the area between the largest and the part below it for the second stage, and the area above the smallest part is for the third stage. This piston has been in severe service and the lower rings are hard to see because they are badly carbonized. The connecting rod is a small replica of the main connecting rod, wrist-pin lubrication being secured in the same way.

The cylinders, however, are lubricated by non-circulating mechanical oilers, the oil being used only once. There are two oilers, one having four plungers and the other two. There is one plunger for each stage of each cylinder. Each of these plungers in one stroke displaces from about 1 to 2½ cubic inches of oil, but in operation over an hour is required to make one stroke—a rachet, worm and screw thread being made use of to reduce the rate of the drive. This is practically all the oil the compressor gets in compressing 450 cubic feet of free air per minute to a pressure over 1,300 lbs. per square inch. Four hundred and fifty cubic feet is over double the quantity of air used in running, and the volume is normally reduced by throttling the suction to the 1st stage. The large reserve is necessary to fill the starting air bottles and for the ship's tanks. The valves are automatic, and, as can be seen in Fig. 16, consist of thin flat steel rings. The instructions are to clean the pistons and rings every six

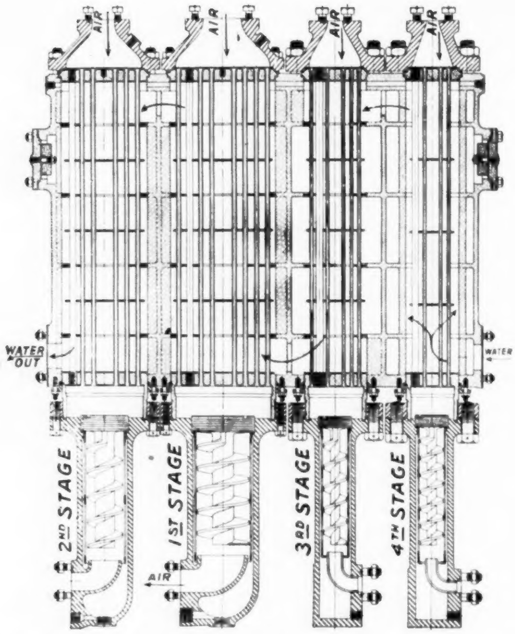

Fig. 17.—Air compressor **intercoolers and separators**. Air at about 350° F. passes down through ¼" I. D. copper tubes in the coolers for the first two stages, and ⅛" I. D. tubes for the other two. The baffling for the water is similar to that for the oil of the oil cooler. The cooling areas are 31, 20, 10 and 5¾ square feet for the 1st, 2nd, 3rd, and 4th stages respectively. The cooling water rises from about 79° to 86° F. in cooling the air to 90° F. The whole nest of tubes is tinned to prevent corrosion.

months, and the valves every 150 hours of service.

Particular precautions have been taken to separate out all water and any oil which are condensed in the intercoolers through which the air passes after leaving each stage. Much more water is separated out in the intercoolers than is commonly realized. With saturated air at 80° F. when compressing full capacity, over 5 gallons an hour would be separated. In normal operation this is probably about one gallon an hour. Fig. 17 shows a section of the air cooler and separator unit for all four stages. In the separators the air is whirled in passing along a helical passage and centrifugal force throws any liquid onto the wall of the surrounding tube. Liquid thus separated passes through narrow vertical slits in the tube and then drains down the outer wall into a retainer provided for the purpose below the air outlet. Every 15 minutes of operation these retainers are "blown down" to remove the condensate which has collected. Notice that a nest of cooling tubes and a separator forms a unit suspended from the top header only, and leaves the tubes free to expand and contract and in this respect is similar to the oil cooler. The lower header of the cooler is made water tight by a gland packed with a rubber ring. These separators are simple and effective, and are an important item in the reliable operation of the compressor. Passage of water into the compressor cylinders greatly interferes with their lubrication, and thus the reliability of the whole unit.

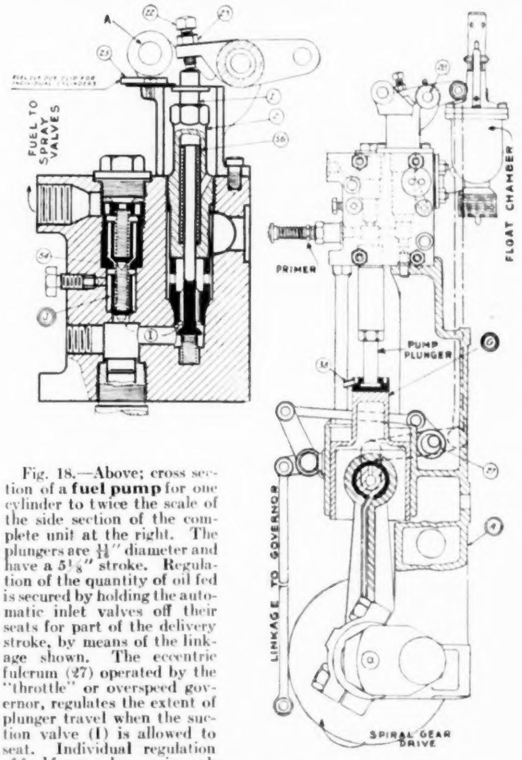

Fig. 18.—Above; cross section of a **fuel pump** for one cylinder to twice the scale of the side section of the complete unit at the right. The plungers are $\frac{11}{16}$" diameter and have a 5⅛" stroke. Regulation of the quantity of oil fed is secured by holding the automatic inlet valves off their seats for part of the delivery stroke, by means of the linkage shown. The eccentric fulcrum (27) operated by the "throttle" or overspeed governor, regulates the extent of plunger travel when the suction valve (1) is allowed to seat. Individual regulation of fuel for equal power in each cylinder may be adjusted by the tappet (22). When the engine control levers are in the "stop" position all delivery is stopped by the rotation of a rod at "A" which opens all suction valves by short arms which bear down on the collars of (1). Any cylinder can be cut out of service by a clip (25) which when set as shown holds the suction valve permanently off its seat.

FUEL SYSTEM

The fuel used in submarine Diesel engines in the U. S. Navy has a viscosity normally of less than 100" Furol* at 77° F., though oil of 1,000" at the same temperature (about 16° Be. gravity) has been burned so satisfactorily when preheated that it is believed still heavier fuel can be used in these engines. The only apparent drawback was that the maximum power developed without a smoky

*Saybolt Furol Viscosimeter has been adopted by the U. S. Government for fuel oils. See LUBRICATION, May, 1921, Page 5.

exhaust was a little lower, possibly because the fuel was not so thoroughly atomized. These engines give a clean exhaust over a surprisingly wide range of speed and load, even at a mere idling speed. In order to accomplish this extreme flexibility several so-called mechanical "complications" have been added, notably means for varying the spray-air pressure and the spray valve lift with the engine power. These features can be called "complications," only in a service where their advantages would be unnecessary, such as for low speed freighters. The fuel system, being at least as novel as some details of the lubricating systems, merits special consideration.

Fuel oil is displaced from the main tanks to a service tank by the admission of sea water. An electric device indicates when the water level reaches a pre-determined height so that the engine can be switched over to another tank, or stopped before salt water is fed to the fuel spray valves. This is not only a mechanically simple method, but eliminates any appreciable change in ballast.

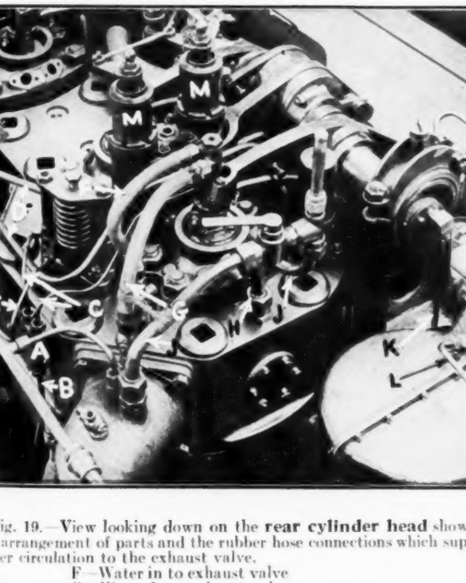

Fig. 19.—View looking down on the **rear cylinder head** showing the arrangement of parts and the rubber hose connections which supply water circulation to the exhaust valve.

F—Water in to exhaust valve
G—Water from exhaust valve
H—Water to exhaust valve cage
J—Water from cage.

All the cylinder cooling water passes through these two paths in parallel and then into the water jacket around the exhaust pipe.

D—Spray air lines to fuel spray valves M
C—Separate Fuel lines to each spray valve from test screw A
B—Single fuel line from fuel pump to test screw A
K—Lubricating Oil Lead to valve rocker shaft
L—Lubricating Oil Lead to spiral cam-shaft gears.
E—Air line to cylinder compression relief valve.

Notice the eccentric mounting of the valve-rocker shaft by which the rockers are lifted from their cams when the cam-shaft is moved to change from the ahead cams to the astern cams.

The service tank is large enough to contain about two hours' supply of fuel and is provided with muslin filter cloths. These are very satisfactory but must be renewed at least every six weeks, and oftener with some grades of fuel. The tank, as well as all fuel oil leads, are recommended to be emptied and carefully cleaned with steam twice a year if possible, as is good practice with any fuel oil system. From the service tank the fuel flows by gravity through a $1\frac{1}{8}''$ I. D. pipe, to a float chamber similar to but larger than those used for automobile carburetors. This float is located just back of the fuel pump, as shown in Fig. 18, and is provided with a sight glass to show the presence of fuel.

The fuel pump unit consists of two groups of four individual pumps, the plungers of each group being driven at 180° from the other. The pump runs at half engine speed, of course.

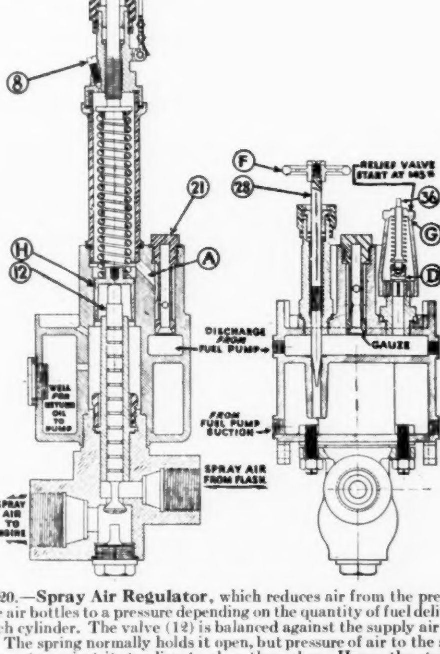

Fig. 20.—**Spray Air Regulator**, which reduces air from the pressure in the air bottles to a pressure depending on the quantity of fuel delivered to each cylinder. The valve (12) is balanced against the supply air pressure. The spring normally holds it open, but pressure of air to the spray valves acts against it, tending to close the valve. Hence the stronger the spring pressure the higher will be the reduced air pressure. Spring pressure is adjusted by hand-wheel E for an air pressure of 600 lbs. when the engine is not running. Lubricating oil from the two extra plungers of the fuel pump enters the space around the spring by passing through a gauze strainer (21) and port A. Exerting a pressure against piston H augments the spring pressure and thus raises the spray air pressure. Control of the oil pressure on H is effected by the by-pass valve (28), all by-passed oil returning to the reservoir shown in the casing, from which it flows back to the pump. A safety valve at the right protects the system against excessive pressure if valve (28) is accidentally closed too far.

Three of the plungers in one group deliver fuel to spray valves while the fourth pumps lubricating oil to the spray air regulator. The three fuel plungers deliver simultaneously to cylinders 1, 5 and 4, while three of the other group deliver to cylinders 6, 2 and 3. The pump is so timed that the delivery stroke of the 1. 5, 4 group is just completed when the piston of No. 6 main engine cylinder is on its firing dead center. All plungers are in a row, are hardened, ground and lapped in their guides so that no stuffing boxes are used, and are loosely coupled to their respective cross-heads so that they will not bind even if not in exact alignment. The small amount of leakage past the plungers, is caught in a trough and drained so that it will not make the engine look untidy. Two check valves in tandem are used for the discharge valves from each pump to securely withstand the very high pressure of the injection air. The quantity of fuel delivered to each cylinder, and thus the engine torque, is regulated in the usual Diesel way by holding the suction valves open for a greater or less part of the stroke. Accurate control of the fuel to each cylinder is very important, because the quantity of fuel burned on one firing stroke is very small. Even in an engine of this size the maximum delivery on a single stroke will be but 0.2 cubic inches which may be visualized as a sphere less than $\frac{3}{4}''$ in diameter.

For the purpose of filling the fuel lines before starting, there is a priming plunger (16) for each fuel pump. When it is operated it makes use of the same valves that are normally used by the regular plunger, and consequently when priming, the throttle must be set in its full open position and the interlock to the starting lever must be disengaged so that the suction valves of the fuel pump will not be held off their seats. When priming, little "test-screws," shown at A in Fig. 19 near the spray valves are opened and fuel is pumped by the primer until it flows out. Drain tubes are provided to carry off the fuel bled out. A check valve is used between these test-screws and the spray valve to prevent passage of spray air when the screw is open. When the fuel lines are filled to these screws, the engine is ready to start as far as the fuel is concerned.

The fourth pump of the fuel pump unit previously mentioned is almost identical to the other pumps. It delivers lubricating oil to the spray air regulator valve at the same rate that fuel is pumped to the cylinders. This regulator, shown in Fig. 20, automatically reduces air pressure from the air flasks to a pressure varying from 600 lbs. per square inch at no load to 1,175 lbs. at maximum load. The more fuel pumped to each cylinder, the higher will the

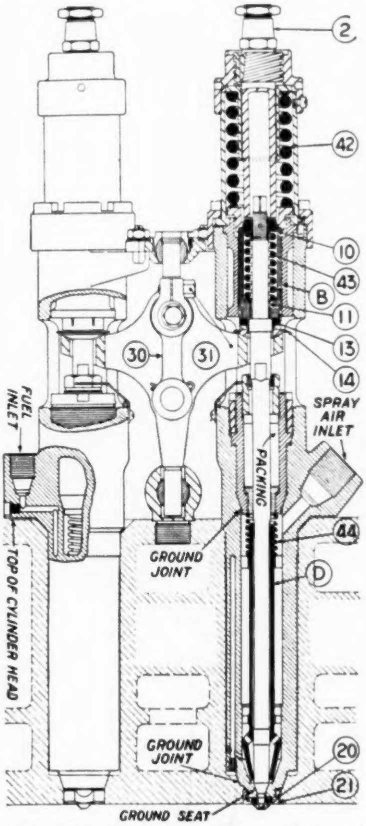

Fig. 21.—**The Twin Fuel Spray Valve.** Notice the extreme simplicity of the atomizer D, in which a few serrations at the bottom serve the purpose of the fine holes and intricate passages used in many other engines to get satisfactory atomization. When the valve (11) opens, air pressure forces a charge of fuel, previously fed by the fuel pump, through the serrations and into the air stream blowing by from the large holes in the atomizer above the oil level. The two valves are synchronized by adjustment of the thickness of washers (13).

Control of the valve lift; two springs act on the valve stem (11); a large one (42) tending to hold it closed by means of a sleeve (B) bearing down on an enlargement of the valve stem but not fixed to it—and a small spring (43) tending to raise it by acting up against the collar (10) screwed to the valve stem. The valve is opened by an upward movement of yoke (31) which in lifting the sleeve (B) removes the pressure of the large spring from the shoulder of the valve stem. The only force remaining on the valve stem is then the small spring. Consequently the valve lifts until the valve stem collar (10) strikes the lower end of the screw (2), which prevents further motion of the valve. Continued lifting of the yoke simply compresses the two springs. The extent of lift is thus limited by the height of the screw (2) which by means of an arm fitted on its tapered upper end, is coupled to all other spray valve arms and a linkage which is moved by the "throttle" hand-wheel. See Figs. 1, 2 and 3.

spray air pressure be. If the engine were suddenly stopped while running at full load, the spray air lines would remain filled with air at the maximum pressure. In order to prevent the existence of such a pressure when starting again, a simple spring loaded relief valve is opened by the control levers when moved to the stop position. This valve is set to blow down the lines to 600 lbs., so the proper pressure for starting is secured automatically.

Fuel Injection Valves

One of the most interesting features of this engine is the use of two spray valves for each cylinder. This is probably done in order to allow larger inlet and exhaust valves than would be possible if space were occupied in the center of the cylinder head by the spray valve and to permit more uniform water circulation. At the same time good combustion is augmented by injecting fuel at two points rather than one. The general arrangement of the twin valves is shown in Fig. 2, together with the single rocker by which they are opened simultaneously. Greater detail is given in Fig. 21. It will be seen to be in several respects similar to the much simpler spray valve shown in Fig. 11 of LUBRICATION for September, 1921. In reality Fig. 21 shows the type of valve used in the 1,750 and 3,000 H.P. engines instead of the earlier 1,200 H.P. type. The difference is primarily in the atomizer.

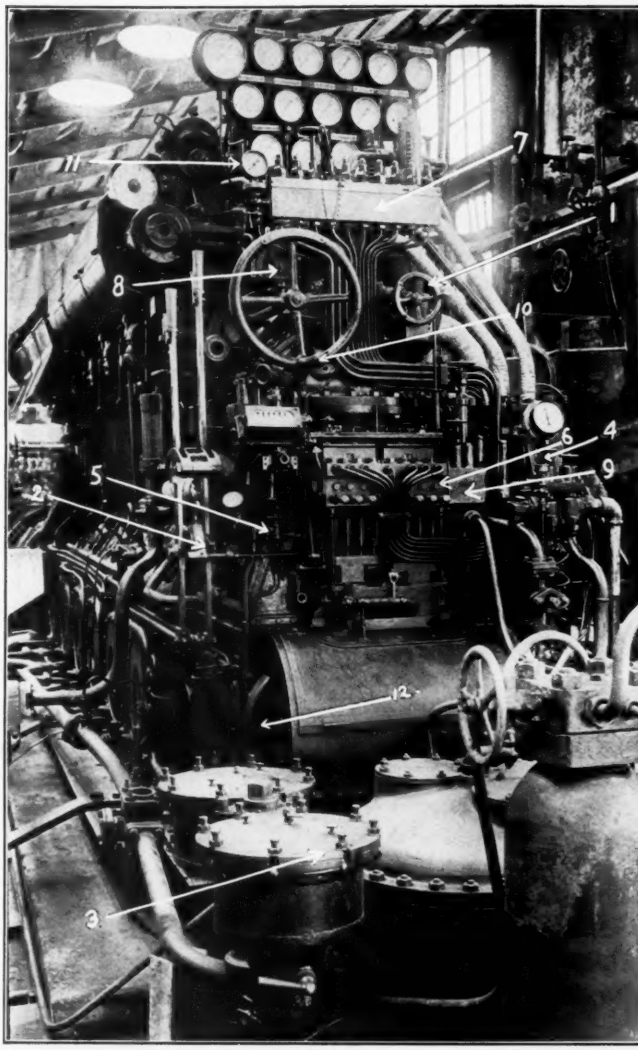

Fig. 22.—**Control end of the Engine**, showing location of many of the parts described in this article. At the left are the two starting levers; just to the right of them is the reverse wheel (10) and the "throttle" control (1). In the submarine a platform is located in a suitable position for the operator to reach these controls, and observe the many gauges above the engine. In the foreground are shown the duplex oil filter (3) of Fig. 7, and immediately to the right is the top of the oil cooler of Fig. 8. The lubricating oil valve of Fig. 12 is shown at (4); the fuel pump unit of Fig. 18 at (6); the spray air pressure reducing and regulating valve of Fig. 20 at (9); and (7) shows the grouping of the valves by which the air compressor intercoolers are blown down and connections made for charging air bottles, etc.

The fuel supply for each cylinder is divided at the test-screw (A of Fig. 19) into a line for each of the two spray valves. Equal division of the fuel is accomplished by: (1) Forcing the oil into each line through a small hole which causes considerable pressure drop; (2) The combined areas of the divided lines are equal to the single supply line; (3) The lines to the valves are equal in length.

When running at low speed the actual time a fuel valve is open is ordinarily greater than at full speed, for the cam holds it open a given number of degrees of crankshaft rotation. The result is that at low speed unless some precautions are taken to the contrary, much more spray air enters the cylinder than is necessary for atomization, which is not only wasteful of the air supply but tends to chill the cylinder charge appreciably. Since great flexibility is needed by these engines, the opening of the

fuel valve is changed by varying its lift, in addition to varying the spray air pressure.

ENGINE CONTROL

Engine control is very simple and almost fool-proof, one hand-wheel ("throttle") automatically regulating not only the quantity of fuel delivered to each cylinder and the lift of the spray valve, but indirectly varying the spray air pressure also. To provide complete control there are in addition only the two starting levers and a large hand-wheel for reverse. The large hand-wheel moves the cam-shaft to positions for ahead or astern by a method similar to that of Workspoor* except that the eccentric fulcrums for the valve rockers are not tilted, and the cam-shaft is slid axially to change cams, like that of Burmeister & Wain, and Worthington‡. In all intermediate positions of this wheel, air operated compression release valves are opened for all cylinders. It is interlocked with the starting levers so that it can be moved only when the latter are set for "stop." On larger engines of this type the reverse mechanism is hydraulically operated and requires but two seconds for complete motion. The longer of the two starting levers when pulled back carries the shorter one with it, and by eccentric fulcrums disengages the spray valve rockers and brings the air starting valve rocker in contact with its cam. Further movement back opens the pilot valve throttle, throwing starting air onto the starting valves. When the engine makes about 100 r.p.m. the long lever is pushed all the way forward, removing air starting valve rockers for cylinders 4, 5 and 6 from their cams, and returning the spray valve rockers for these cylinders so that they can start firing. Cylinders 1, 2 and 3 are still air driven, and after a few more revolutions they are switched over also, by pushing the short lever forward. In stopping, the long lever is pulled half-way back and draws the other with it, shutting off all fuel irrespective of the throttle position, and blows down the spray air lines to 600 lbs. The flow of cooling water from the engine pumps, of course, starts and stops with the engine. There is hence no possibility that the operator can incorrectly handle the controls, nor run the engine with any adjustment of spray air pressure or spray valve lift except that for which it was intended. No miss-move can be made even if he "loses his head."

When starting the engine, time must be given for parts to warm up before putting on heavy load. Operators are instructed when starting an engine from cold, not to come to normal cruising speed until it has been running at least five minutes; to full speed, for 15 minutes; nor to emergency speed for 20 minutes. After stopping, circulation of the lubricating system must be continued by the electric auxiliary oil pump for 15 minutes to cool off the pistons. When a long layup is planned the engine should be turned over at about 200 r.p.m. for a quarter of an hour with all lubricating devices in operation to assure all moving parts being thoroughly oiled. All external parts are to be cleaned and thickly covered with grease to prevent rusting. The cleaning, greasing and motoring should be repeated every month, and in addition—as is customary with engines—the crank-shaft should be jacked over by hand once a day, care being taken to stop with the piston in different positions each time.

CONCLUSION

This article has been confined primarily to the lubricating and fuel systems of the engine, but the friction clutch, located between the engine and the electric motor can hardly be passed by without mention, because of the large power transmitted. It is a double cone, iron to iron design, running in just enough oil to prevent sudden seizure. It is very simple, and is readily controlled by a piston operated by oil under the influence of compressed air. It is interesting to note that in extended trials of this engine at the New York Navy Yard, one grade of lubricating oil was used throughout the whole engine; for piston cooling, engine lubrication, air compressor lubrication and even the clutch, except that a little cylinder stock was added. This is a point of particular interest, and should be a strong answer to those who insist that a high flash point cylinder oil which is unsuited either for engine lubrication or piston cooling, must be used for air compressors in order to prevent explosions in the air bottles.

*LUBRICATION, October, 1921, Fig. 27.
‡LUBRICATION, October, 1921, Fig. 26.

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Edgar C. Ruwe Co., Inc.
47 West Street, N. Y. C.